RECONSTRUCTION OF A THRUST BEARING
FOR
A 75 C. F. S. CENTRIFUGAL SEWAGE PUMPING UNIT

BY
L. W. A. BUNGE

ARMOUR INSTITUTE OF TECHNOLOGY
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RECONSTRUCTION OF A THRUST BEARING FOR A 75 C. F. S. CENTRIFUGAL SEWAGE PUMPING UNIT

A THESIS

PRESENTED BY

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TO THE

PRESIDENT AND FACULTY

OF

ARMOUR INSTITUTE OF TECHNOLOGY

FOR THE DEGREE OF

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Abbreviations and Terms

C. F. S.	Cubic feet per second
C. F. M.	Cubic feet per minute
Datum	The zero bench mark of the City of Chicago - About normal lake level
C. C. D.	Chicago City Datum
P. S.	Pumping Station
Invert	Bottom of sewer or conduit
Crown	Top of sewer or conduit
Dia.	Diameter
I. D.	Inside diameter
O. D.	Outside diameter
C. I.	Cast Iron
H. P.	Horse-power or High Pressure
I. P.	Intermediate Pressure
L. P.	Low Pressure
K. W.	Kilowatt

INTRODUCTION

The object of this thesis is the description and analysis of changes in the design of a thrust bearing on a 75 C.F.S. centrifugal sewage pumping unit. The reconstruction of this bearing is part of a task of overhauling the entire #2 steam operated pumping unit at the 39th St. Pumping Station (Chicago) of the Sanitary District of Chicago. Although the changes in design of the bearing constitute the kernal of this account, the job of overhauling and the consequent elimination of other troubles, are, exclusive of the alterations here to be discussed, of a comprehensive enough nature to justify a detailed report. Such a report would not be complete, however, without some treatment of the first-mentioned subject, and the latter, having a relative value to it, is therefore touched upon, supplementarily, as briefly as seems wise, consistent with conveyance of adequate information about the former. For similar and equally momentous reasons, the whole station and especially the No. 2 unit are described.

The writer at this time is in the employ of the Sanitary District of Chicago, and his duties include the supervision of several pumping stations. Thus it is that the work mentioned, of overhauling this unit was entrusted to his care and his close touch with it has made possible the opportunity for gathering the information herein given.

(The Sanitary District of Chicago is an Illinois municipal corporation and is supported by part of the tax moneys collected in the territory covered by the "District". Its business is that of generating electricity at the hydro-electric plant at Lockport, Illinois, selling same to consumers; and sewage disposal for the City of Chicago and adjacent localities).

Credit is hereby acknowledged and gratitude expressed for help and support given the writer as follows:

Mrs. Julia Beverage, Librarian, Armour Institute of Technology - Assistance in accumulating engineering data.

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THE STATION

In order that the necessity of continuous operation of the pump and therefore the importance of reliable design of anything entering into a vital part of it, may not be underestimated, the entire station together with the work expected of a sewage pump, is described.

The 39th St. Pumping Station is located on the bank of Lake Michigan at the foot of 39th St. - Chicago. Its purpose is to pump the sewage from the low level of the 39th St. intercepting sewer to a high enough level, so that, after it is diluted (at the Station) it will flow through a conduit into the Stock Yards Slip of the Chicago River, from whence it empties into the drainage canal. The territory drained by the station covers about 22 square miles.

The principal equipment of the station consists of two 75 C.F.S. and two 250 C.F.S. triple expansion steam driven, centrifugal sewage pumps; two 667 C.F.S. triple expansion steam driven screw pumps for fresh

water; one 4000 k.w. - 12000v steam turbo-generator with transformers etc.; eight boilers; coal conveyor with crusher in line; overhead suspended coal crane with grab-bucket; and steam-jet ash-handling system. The sewage and screw pumps are products of Allis-Chalmers Manufacturing Company, Milwaukee. The sewage pumps are of the velocity type, each driven by a horizontal corliss engine, the three cylinders of which are set at 120° with each other. (See plates 1 and II and table of Pumping Engine Data). The shaft is vertical. The screw pumps are of the propeller blade type and each is driven by a vertical corliss engine of which the high and intermediate cylinders are in tandem and the low pressure cylinder is connected to a second crank. (See plates 2, 3 and 4.) The shaft is horizontal. All pump engines exhaust into Tomlinson Barometric Condensers. Condenser engines and pumps were made by the American Blower Company. The 4000 k.w. unit is a G. E. generator driven by a Curtis turbine exhausting into a surface condenser. It is used to help carry the peak which

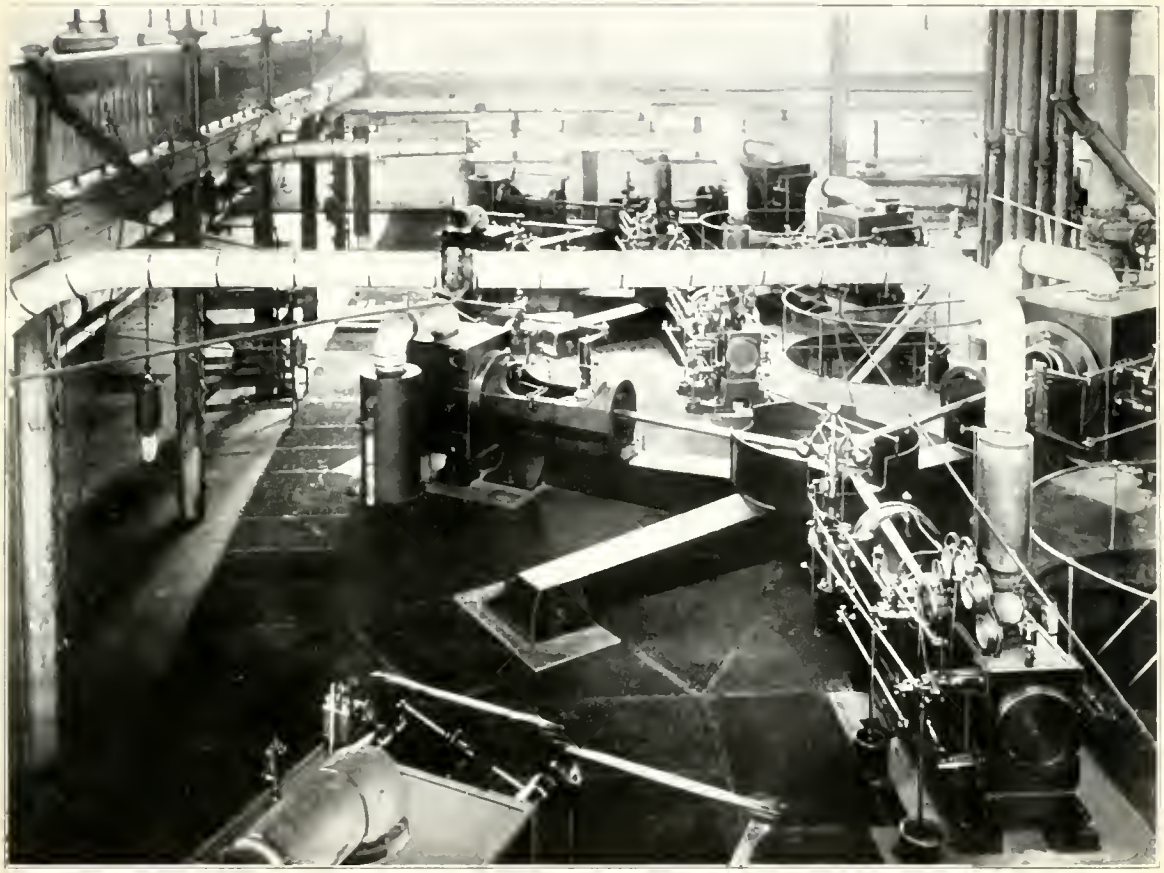

PLATE I

The Sewage Pump Engines

sometimes requires more current than the Hydro-Electric Plant at Lockport can supply. The natural growth of electric consumption made its installation necessary some years after the construction of the Lockport and 39th St. Stations. Boilers 1 to 6 inclusive - rated at 264 H.P. each - are of the Altman & Taylor make with Mansfield chain grates. They operate by natural draft. These were installed when the station was built. Boilers 7 and 8 - rated at 500 H.P. each - are Edge-Moor's with Taylor underfeed Stokers, have superheaters, and use forced draft. These were added a short time after the installation of the turbo-generator in order to meet the increased steam demand. The chimney is 9 ft. in diameter by 188 ft. high above footing, and is made of radial pressed brick with granite block base. (See plate 5).

Coal is delivered from R. R. cars on the siding (See plate 6) to a pit under track, is conveyed to the crusher and is then elevated to a belt running above bunkers over the boilers. A diverting machine

distributes the coal to the proper bunkers from which down-spouts supply the stokers. Storage space is provided by three concrete pits of 300 tons capacity each (heaped over) just north of the plant and under coal crane, and by adjacent ground leased from the Illinois Central R. R. Co. (See Plate 7).

Ashes are loaded into cars directly from the tank of the handling system (American Steam Conveyor Co.), but a stone lined pit of 400 tons capacity, to the north of the station and under coal crane permits of temporary storage in case of car shortage.

Minor equipment includes two 65 k.w. generators, driven by direct connected high speed reciprocating engines; for lighting and power; feed water heaters; boiler feed pumps; stoker auxiliaries, etc.

Plate 8 shows a sectional plan of the engine and pumping room. Sewage enters the station at the south-west corner from A, the 39th St. intercepting sewer. B is the location of the screen chamber (shown in sectional elevation on plate 9) in which aretwo vertical screens 20 ft. wide and 16 ft. high. The

Pumping Engine Data.

39th St. Pumping Station. The Sanitary District of Chicago.

Engine.	Centrifugal No. 1.	Centrifugal No. 2.	Centrifugal No. 3.	Centrifugal No. 4.	Screw. North.	Screw. South.
Rated capacity - C.F.S.	75	75	250	250	667	667
Rated head - Ft.	24	24	7	7	7 Normal. 10 Max.	7 Normal. 10 Max.
H.P. Cylinder diameter - Inches.	12	12	16	16	22	22
I.P. " " "	22	22	30	30	38	38
L.P. " " "	32	32	46	46	62	62
Stroke - Inches.	30	30	30	30	42	42
Impeller diameter - Ft.	7'-3"	7'-3"	9'-3"	9'-3"	14'-9"	14'-9"
Center line of pump below C.C.D.	17'-0"	17'-0"	6'-0"	6'-0"	13'-8"	13'-8"
Top of suction throat below C.C.D.	24'-0"	24'-0"	16'-0"	16'-0"	6'-2¾"	6'-2¾"
Size of suction: Inlet - Ft.	4'x6'	4'x6'	9'x11'	9'x11'	14'-9"	14'-9"
" " " Throat "	4'-0" dia.	4'-0" dia.	7'-0" dia.	7'-0" dia.	14'-9" dia.	14'-9" dia.
" " " At pump "	3'-6" dia.	3'-6" dia.	6'-0" dia.	6'-0" dia.	14'-10½" dia	14'-10½" dia.
Discharge diameter - Ft.	4'-0"	4'-0"	6'-5"	6'-5"	14'-9"	14'-9"
Shaft diameter - Inches.	9" and 9¾"	9" and 9¾"	11¼" and 12"	11¼" and 12"	14"	14"
Speed (max observed) - R.P.M.	120	120	72	72		

L.W.A.B. - May, 1918.

screen frames are made up of three 12 inch horizontal I beams cross connected by two 12 inch vertical channels with diagonal angles for braces. Vertical rectangular bars 2" x 3/4" on 4" centers constitute the mesh of the screen and these are bent back at right angles on the bottom, counter to the flow, in order to catch heavy matter which slides down the screen when lifted for cleaning. One screen should stay down while the other is raised - by chain hoists - above the sewage for this purpose.

The elevation of the invert of the intercepting sewer where it enters the screen chamber is - 19.6 to - 20. That of the bottom of the screen chamber, at that point is - 21. The bottom of the suction chamber G slopes eastward from the screens to elevation - 23, and the width narrows to 12 ft.

The No. 1 and No. 2 units are 75 C.F.S, and No. 3 and No. 4 are 250 C.F.S. The suction inlets are elliptical in shape and measure 4 x 6 ft. for No. 1 and No. 2 and 9 x 11 ft. for No. 3 and No. 4 pumps. Between suction inlets No. 2 and No. 3 is an automatic

swinging gate H, hinged on top, opening eastward. The frame against which the bottom of this gate closes is supported by a concrete wall across the lower part of the suction chambers, so that the sewage, in order to pass this point must rise to elevation - 15. This gate can also be held open by a chain hoist. Units 1, 2, 3 and 4, pump the sewage from the suction chamber (G) into the gravity channel N, whose invert level is - 6. Fresh water- from Lake Michigan comes into the station at intake M and due to the fact that the lake level is generally about one foot higher than that of the river, it flows by gravity through channel N. Here it mixes with the sewage, diluting it, and the entire discharge from the station passes via the conduit V on its way to the river. At such times as the natural flow from the lake does not sufficiently attenuate the sewage, the screw pumps K and L are cut into service. These discharge additional water through channels O and P.

The lift gates, I, J, Q, R, and S, are operated by chain hoists. The swinging lock gates, T, U and

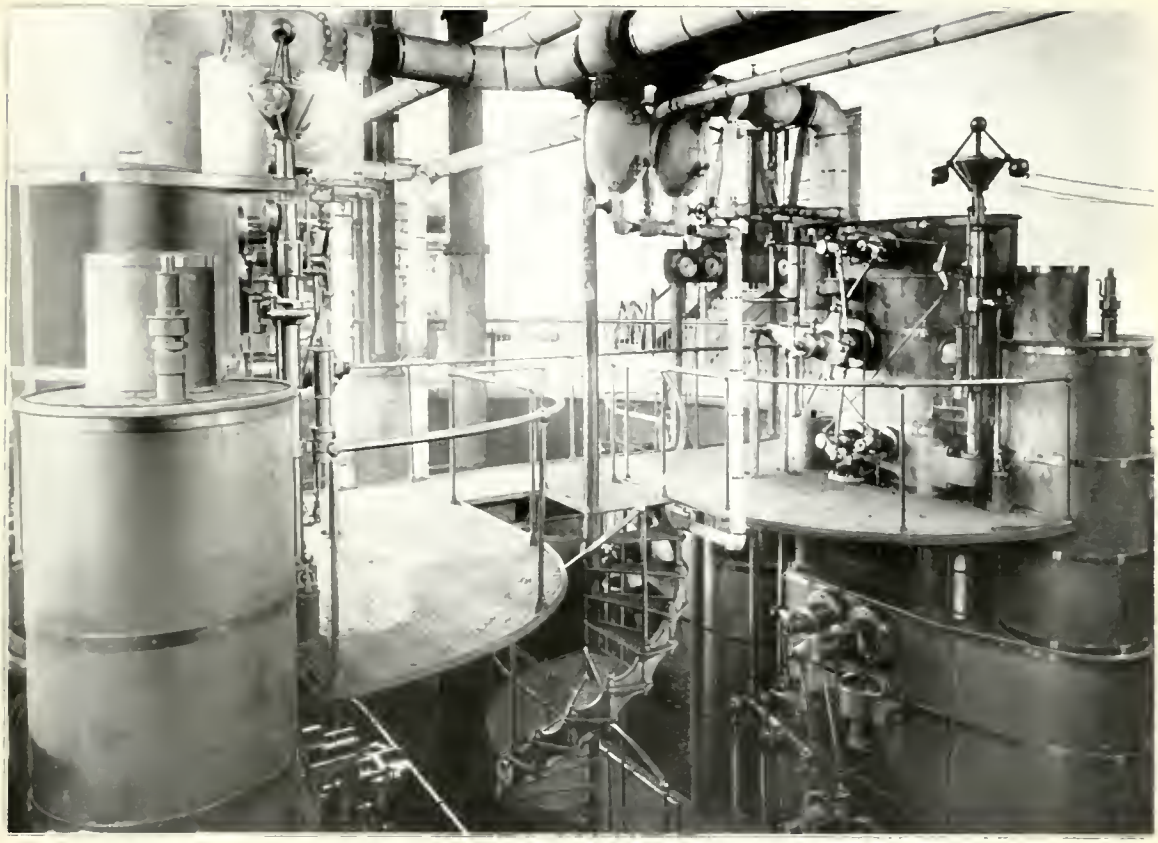

PLATE 2

The Screw Pump Engines

W are actuated by oil cylinders which receive their pressure from small electrically driven pumps. Almost any combination desired can be obtained by proper manipulation of the gates. For example, if there should be more sewage than units 1, 2, 3, and 4 can handle, by closing gates R, S, and W, and opening gates I, J and U, the screw pump L could be used in parallel with the sewage units to pump sewage. (The reader will note that such a predicament would be serious. The few times that it has happened, co-existing condictions have given rise in each case to public investigation). Then again, if disaster should overtake the screw pumps, by closing gates J and W and opening gates I and S fresh water could be pumped by sewage units No. 3 and No. 4. In this case the automatic swing gate H would prevent the lake water from backing up the sewer. Advantage is taken of this latter arrangement also to prime pumps No. 3 and No. 4.

It should be borne in mind that sewage includes not only the waste from the home or factory, but also

PLATE 3

Stationary Guide Vanes and Cone Bearing Housing of Screw Pump during Construction The Buckets of the Impeller very closely resemble these Vanes.

the drainage from streets. Thus when it rains heavily, or when snow thaws rapidly, much more sewage has to be pumped at the station. Failure properly to handle this sewage causes the flooding of basements. Neglect in sufficiently diluting the sewage brings forth a storm of protests from the people, because of the stench in the river. Needless to say that the health of a vast population is at stake, in the operation of this station alone. This is one of several in Chicago.

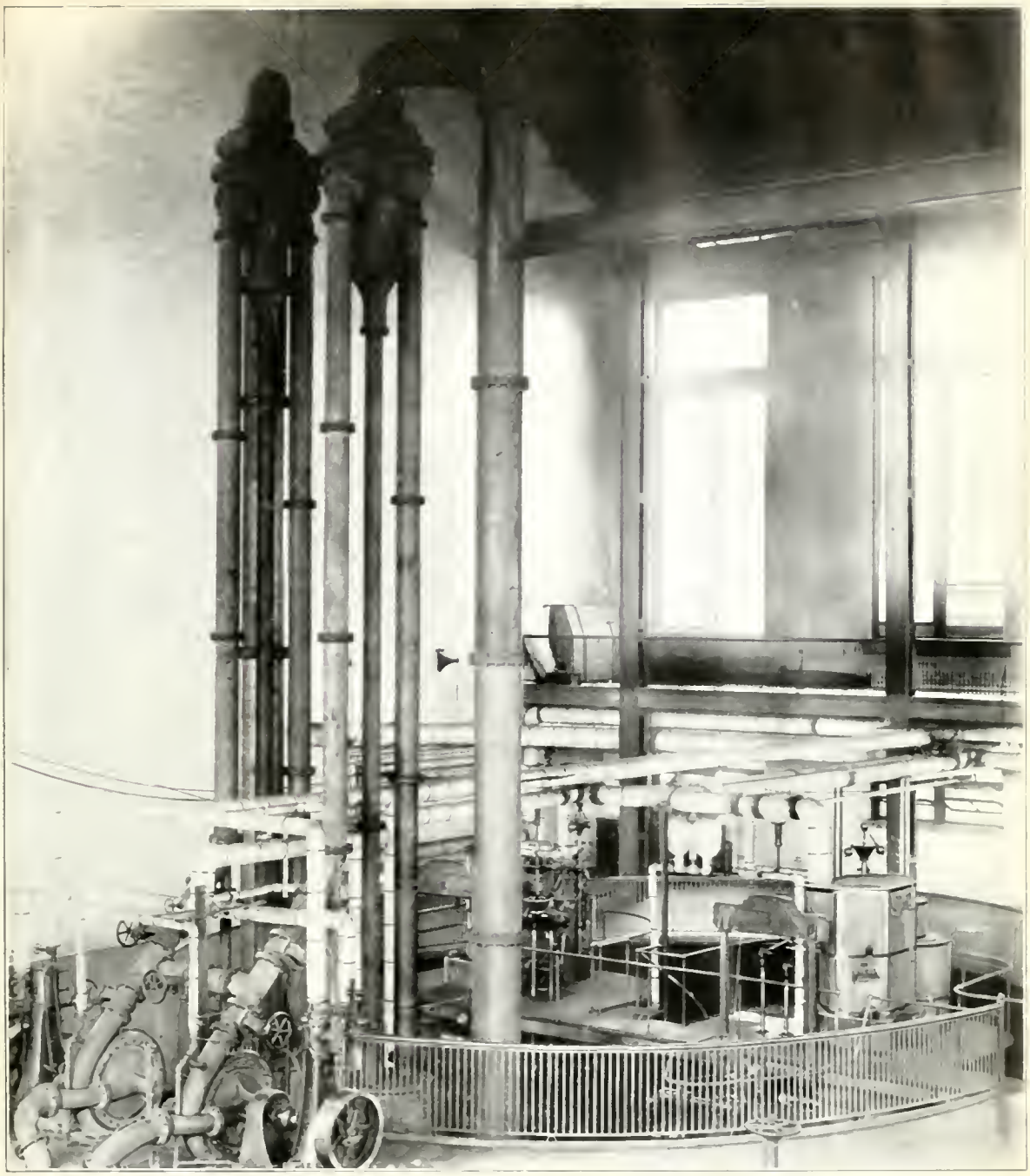

PLATE 4

Tomlinson Condensers for Screw Pump Engines

(Condensed auxiliaries are shown at left,

Screw 46 Pump Engines at Right.)

THE UNIT

As stated before, the unit in question is the #2 sewage pump. It is rated at 75 cubic feet per second capacity against a total head of 24 feet. It was bought under a contract which contained clauses providing a heavy penalty for failure to meet specifications and a liberal bonus for exceeding them. It is not necessary to mention the outcome of the purchase. The overdesign is borne out by the capacity table on page 15 and the set of curves on plate 10, made from the table. This data shows clearly, that the pump was designed for no more than seventy-five per cent efficiency which is good practice - and then at least ten per cent further reduction, to be sure that the pump would come in the bonus class.

The best idea of the unit can be obtained by studying the cut on plate 11 in conjunction with the assembly drawing of plate 12. The crank and cylinders are above floor level; the shaft, thrust bearing, lignum vitae guide bearing and pump are

PLATE 5

The Station as Seen from the Lake

set in a concrete well. The thrust bearing is just above the level of the oiling gallery. Part of the engine room floor and well wall are broken away in the cut, to give a complete view of the unit. As one might surmise from the table and curves mentioned above, the pump can be run at various speeds to suit the load.

Periodically conditions make it apparent that one or more of the units is in need of a thorough overhauling. The last time the No. 2 unit was overhauled was about five years ago. At the present time the need of attention to the innermost parts of the pump was suggested by a lack of capacity and by an unbalanced action while running. The former was indicated by the fact that the level of the sewage in the suction chamber would not lower fast enough at such times as it was known that the flow was not as great as the rated capacity of the pump. The latter could be seen by looking at the engine while it was running. It was also noticed that at times of heavy rainfall the combination of the four

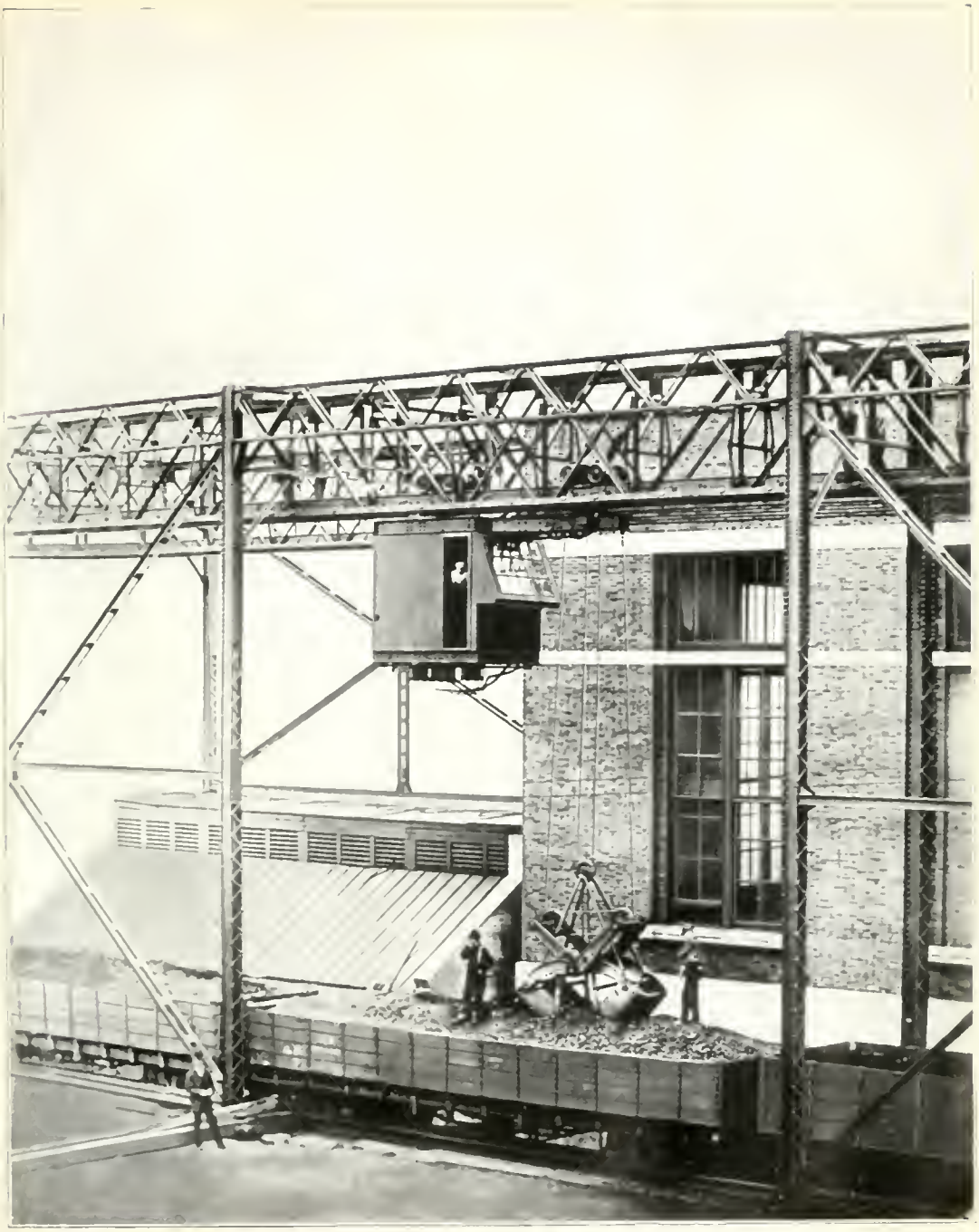

PLATE 6

The Coal Crane

Table of Pump Capacities.

75 C.F.S. Centrifugal Sewage Pump. 39th St. Pumping Station.

The Sanitary District of Chicago.

Speed	60 R.P.M.		65 R.P.M.		70 R.P.M.		75 R.P.M.		80 R.P.M.		85 R.P.M.		90 R.P.M.		95 R.P.M.		100 R.P.M.		105 R.P.M.		110 R.P.M.		115 R.P.M.		120 R.P.M.		125 R.P.M.		Speed
Head	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	C.F.M.	C.F.S.	Head
6 Ft.	3240	54	4140	69																									6 Ft.
7 Ft.	2520	42	3600	60	4560	76	5340	89	6060	101	6780	113	7320	122															7 Ft.
8 Ft.	1620	27	3000	50	4020	67	4860	81	5580	93	6360	106	6980	116	7620	127													8 Ft.
9 Ft	540	9	2220	37	3480	58	4440	74	5160	86	6000	100	6660	111	7290	121.5	7920	132	8700	145									9 Ft.
10 Ft.					2880	48	3900	65	4740	79	5580	93	6240	104	6960	116	7560	126	8340	139									10 Ft.
11 Ft.					2160	36	3360	56	4260	71	5160	86	5880	98	6600	110	7260	121	8040	134									11 Ft.
12 Ft.					1260	21	2760	46	3720	62	4740	79	5460	91	6240	104	6960	116	7680	128									12 Ft.
13 Ft.									3240	54	4260	71	5100	85	5880	98	6600	110	7380	123									13 Ft.
14 Ft.									2640	44	3780	63	4680	78	5520	92	6240	104	7020	117	7800	130	8550	142.5					14 Ft.
15 Ft.									1920	32	3240	54	4200	70	5100	85	5880	98	6720	112	7500	125	8220	137	8910	148.5	9630	160.5	15 Ft.
16 Ft.									1140	19	2700	45	3720	62	4710	78.5	5520	92	6360	106	7200	120	7920	132	8610	143.5	9310	155.2	16 Ft.
17 Ft.															4860	71.	5160	86	6000	100	6870	114.5	7620	127	8280	138	9000	150	17 Ft.
18 Ft.																	4800	80	5640	94	6540	109	7290	121.5	7980	133	8750	145.8	18 Ft.
19 Ft.																									7680	128	8400	140	19 Ft.

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sewage units, seemed to lack capacity. An investigation led to tests with consequent recommendations for all pumps. The principle tests of interest were those of capacity and were made by the salt method. The recommendations called for the more or less complete overhauling of all four sewage units. Proper authority was granted and the work begun. No. 1 unit was gone over most minutely, worn parts trued up or replaced, and reassembled in good order, along the lines, throughout, of the original design. While the work on No. 1 was in progress the engineer directing this work, before the writer, studied the weaknesses of the entire unit, with a view of making permanent corrections in No. 2. The most serious trouble, and the one which, if not checked up occasionally, might prove most disastrous was in the thrust bearing.

The principle recommendations set forth in connection with No. 2 unit are as follows:

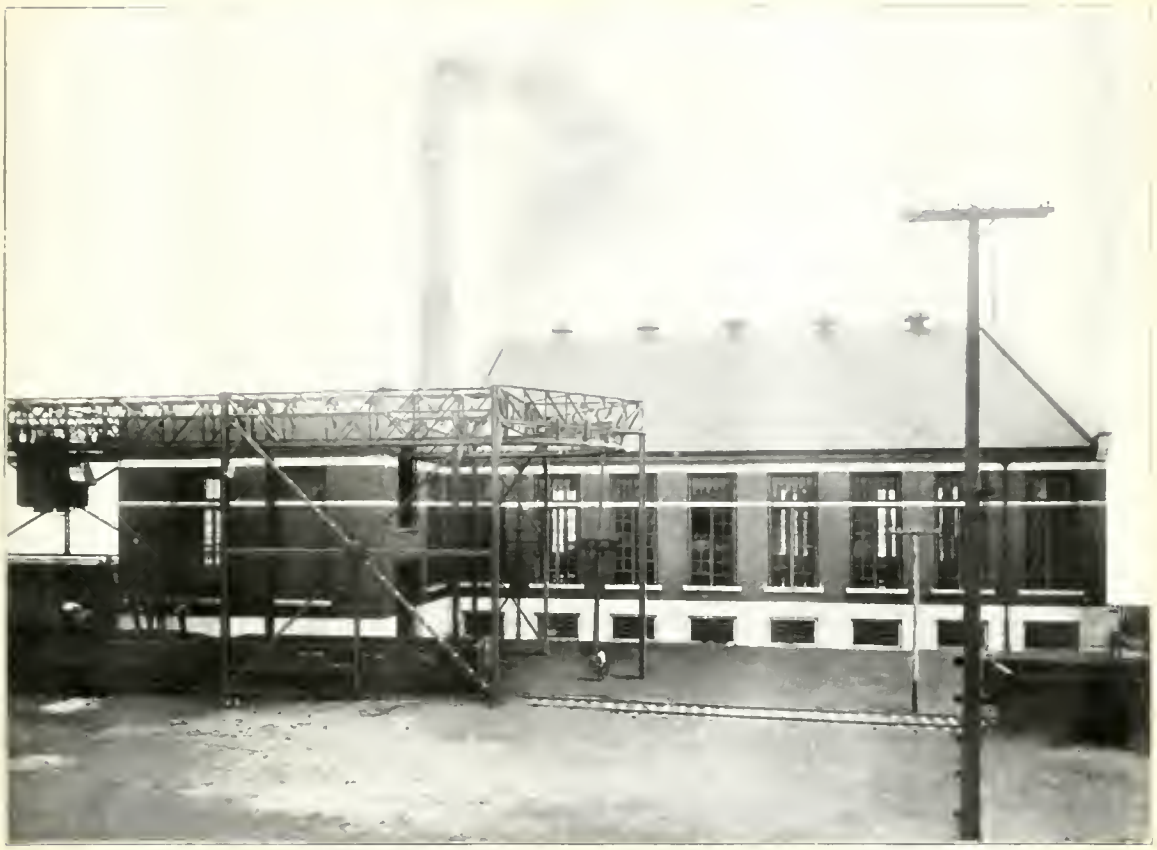


PLATE 7

The Station, looking East.

Ground West of Track is used for Coal Storage.

Part of Crane Structure is also shown.

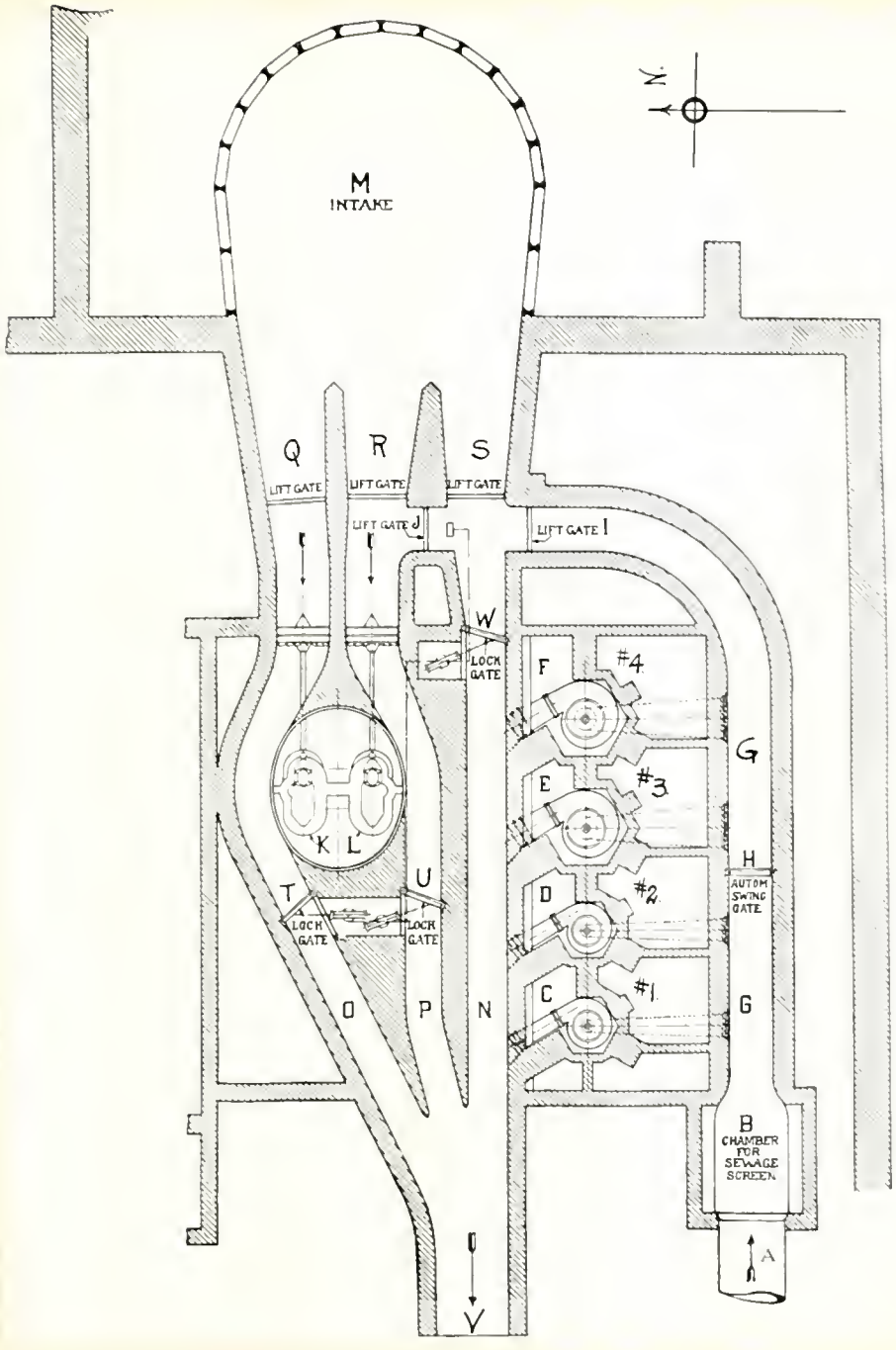
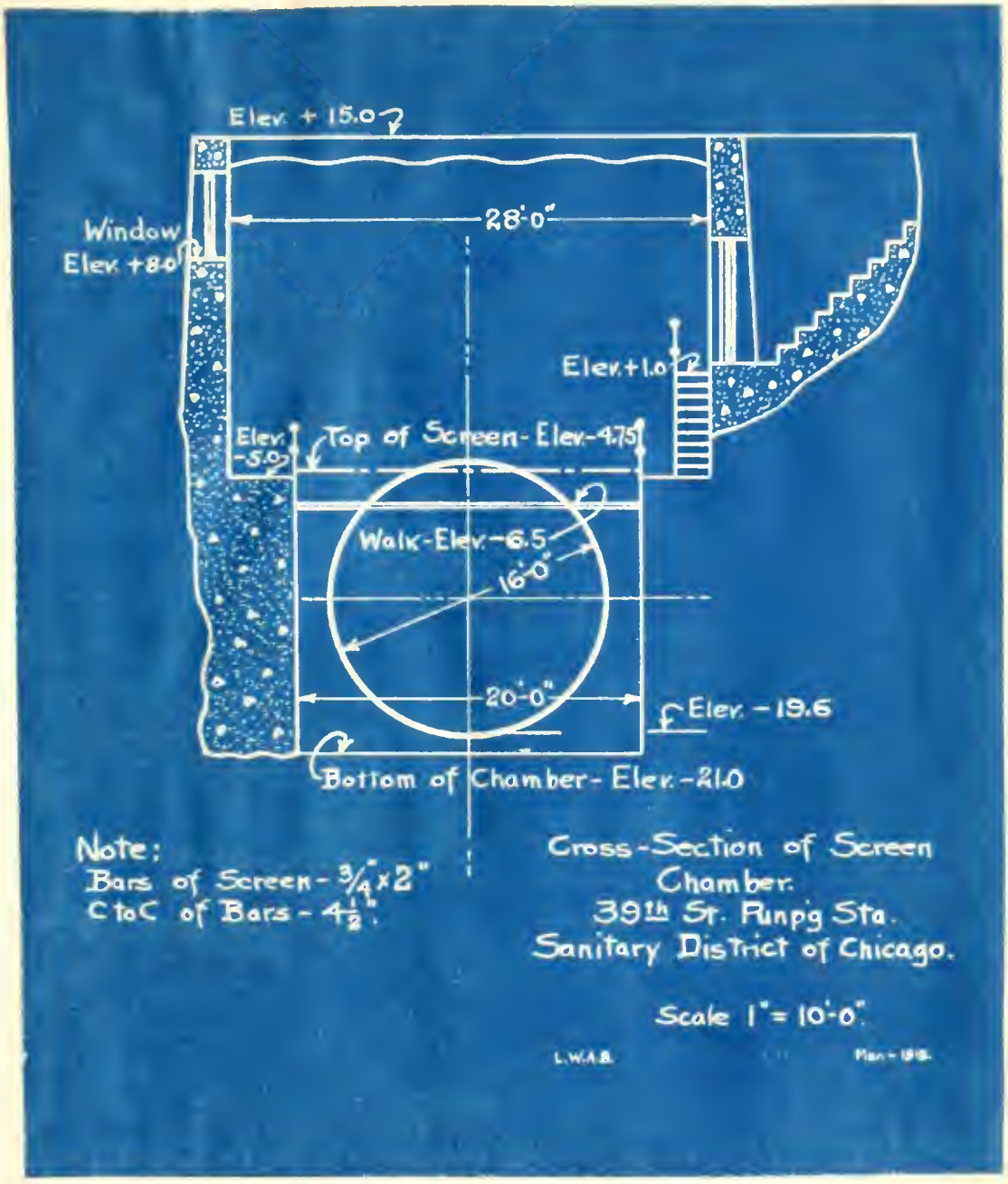


PLATE 8

Sectional Plan of Engine and Pump Room.

1. Reboring steam cylinders where needed
2. Rebabbiting and fitting worn bearings
3. Reconstructing cross-heads so that adjustments could be made for taking up wear. (Formerly there were no such adjustments).
4. Putting thrust bearing in working order
5. Installing new Lignum Vitae bearing and making new journal for same.
6. Truing up impeller.

Of special interest in connection with the reconstruction of the thrust bearing are items 5 and 6. The lignum vitae bearing is made up of a brass casing into which are dove-tailed lignum vitae strips. The strips form the bearing for the shaft just above the pump proper (See plate 12). This bearing had been allowed to become so badly worn that the shaft came in contact with metal, becoming grooved to such an extent that it was necessary to machine it down and press on a bronze sleeve to form a new journal.



PIATE 9

Sectional Elevation of Screen Chamber.

This play permitted a whipping action in the shaft which caused the clearance faces of the impeller to become worn in spots. These faces had to be trued up in the lathe. This work was done at the shops of the Allis-Chalmers Manufacturing Company, Milwaukee. When the runner was in the lathe being cleaned up, a Y shaped crack was discovered in it. This gave entrance for water into some of the cored spaces. Other cored spaces remained tight. This made the impeller heavier on one side than on the other and contributed largely to the unbalanced action of the pump. Upon opening the pump in the first place several large pieces of wood were found in the impeller. These could not get out. The original recommendations intimated a suspicion of the presence of such foreign matter, and gave that as the cause of the unbalanced action. Although not to be disregarded it can readily be seen that this was a less important factor.

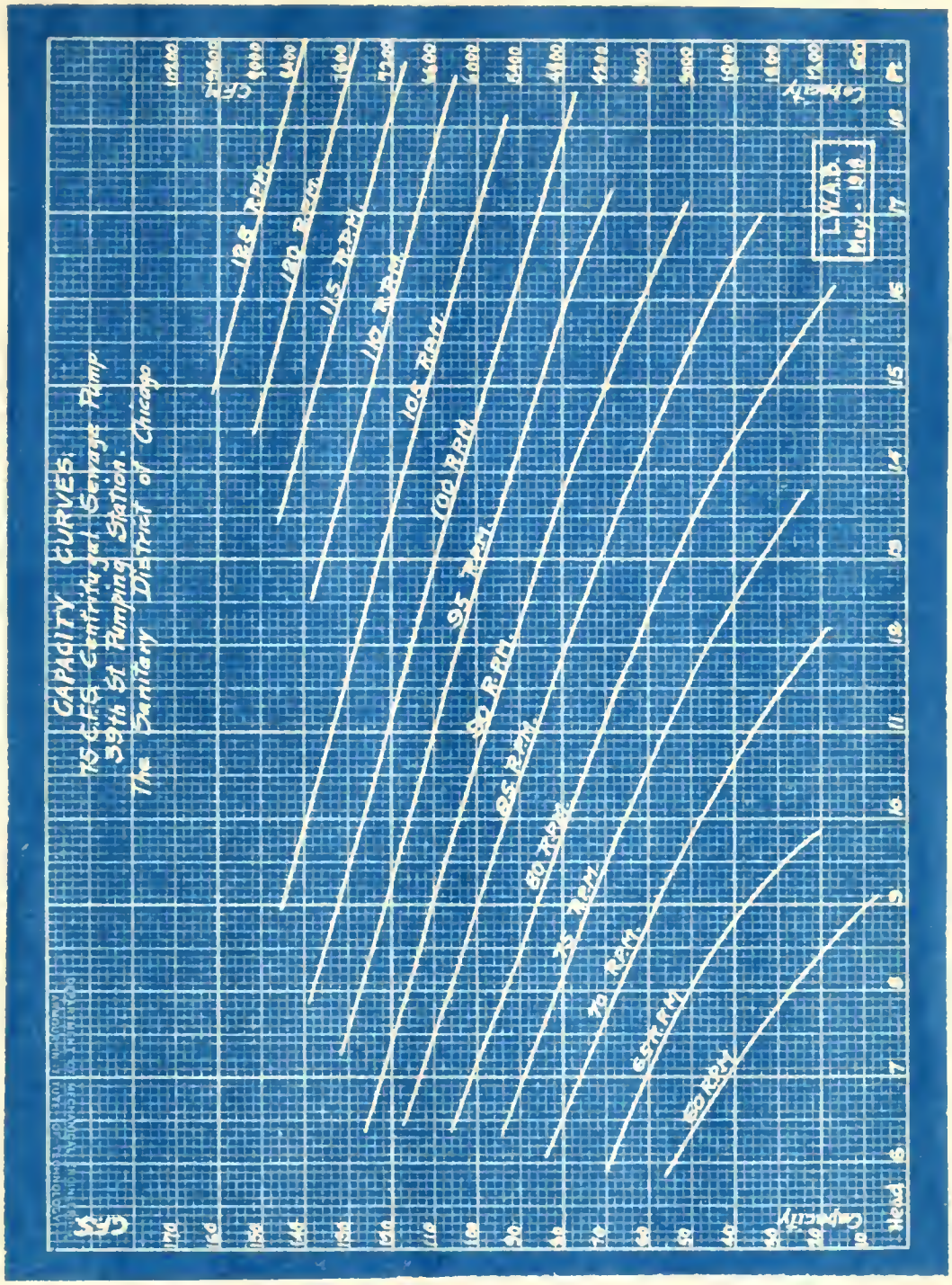


PLATE 10

Capacity Curves for 75 C.F.S. Centrifugal Pump.

THE ORIGINAL BEARING AND ITS WEAKNESSES

The thrust bearing carries the entire weight of the shaft and impeller and the hydraulic thrust due to lifting the water at the suction against a head. The bearing as originally designed and installed is shown on Plate 13.

The movable half of the bearing consists of a split nut whose threads fit the threads of the shaft to facilitate vertical adjustment. The bolts which hold the two parts of the nut together are capable of tightening it on the shaft sufficiently to cause it to rotate as one piece with it.

The stationary half of the bearing is made up of two castings, one resting within the other on a spherical face (Marked "Face" on drawing). The outer casting is rigidly fastened to the surrounding concrete and steel framing. A dog or shoulder in the outside casting, engaging a groove on the inner casting prevents the latter from revolving but permits it to have a rotating motion about axes perpendicular to the shaft, to compensate for slight

disalignments of the shaft.

Between the revolving and stationary halves of the bearing are four alternate bronze and steel plates.

The top plate is bronze and turns as one piece with the split nut or movable half of the bearing. Its connection to the split nut is by means of dowels. The bottom or lower plate is steel and being doweled to the stationary half of the bearing has no motion. The dowels serve the further purpose of preserving radial alignment of these two plates.

The object of the two intermediate plates is to spread the bearing surface over greater area. When in good working order the angular velocity of any plate would be less than that of the one above it. It is obvious, however, thatanything, causing more friction at the contact surfaces of any two plates, than at the other two pairs of contact surfaces, would throw the whole burden - as a bearing - upon these other two pairs of con-

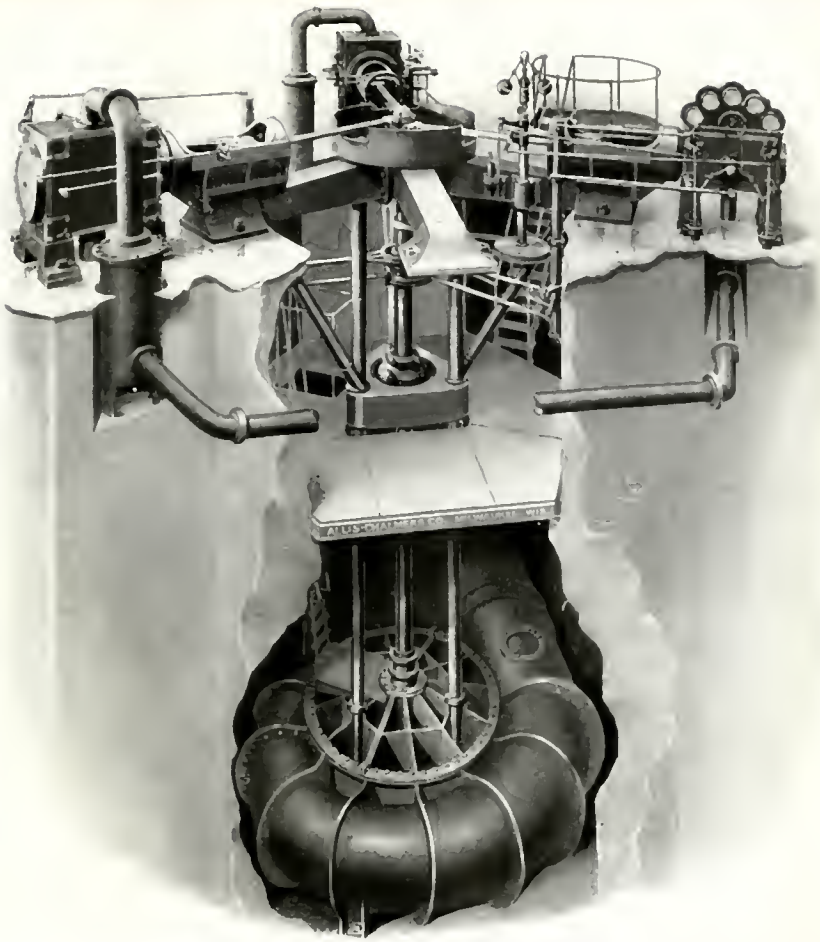


PLATE 11

The 75 C.F.S. Sewage Pumping Unit.

tact surfaces. Similarly it can be imagined that the entire difference in motion between the shaft and the stationary half of the bearing could exist at a single pair of contact surfaces. Therefore, it is essential that the area of these plates is sufficient to take care of just such a condition. That this has actually been considered in the design is evident from the bearing pressure shown in the following rough commercial calculations:

PRESSURES ON OLD TYPE BEARING

W = Load on bearing including weight of moving parts, hydraulic thrust, etc., approximately 20 ton = 40,000 lbs.

d_1 = Outside diameter plate 2'- 3 1/4" = 27 1/4" (effective)

d_2 = Inside diameter plate 12 1/4"

A = Area plate = $\frac{\pi}{4}\left(d_1^2-d_2^2\right) = \frac{\pi}{4}\left[(27\ 1/4)^2 - (12\ 1/4)^2\right]$ = 465.35 square inches.

f = Average unit pressure = lbs. per sq.in.

$f = \frac{W}{A} = \frac{40,000}{465.35}$ = 85.96 lbs. per sq. in.

These values are low for thrust bearings of this sort. In practice, pressures of 150 lbs. per sq. in. are common.

Radial alignment of the intermediate plates is maintained by steel and bronze guide rings (also shown on plate 13). The steel ring guides the bronze plate and the bronze ring guides the steel plate. These rings are machined to a press fit on the collar which is integral with the inside casting of the stationary half of the bearing.

Lubrication is accomplished as follows: From plate 13 it can be seen that the upper part of the inside casting of the stationary half of the bearing forms a cup which resembles an inverted doughnut cutter. All plates are below the lowest level at which oil in this cup could overflow. Oil flows toward the shaft through radial grooves in the under side of the bottom plate. At the collar it rises through oil holes in the guide rings from whence it is distributed centrifugally over the friction areas by means of tangential grooves in the

contact surfaces of the plates. Surplus oil at the collar is induced away centrifugally by the 1/4 x 1/8 oil grooves in each face of the split nut. A glass standpipe serves as an oil gauge and an inspection door in the hood, which covers the entire mechanism, affords an opening for refilling and for feeling the temperature of the oil with the hand. A drain is also shown, on the drawing, to empty the system for cleaning and replenishing.

To the lubricating system belongs also the cooling arrangement. Plate 13 shows a cored space in the lower part of the inside casting of the stationary half of the bearing. This space forms a jacket through which water is circulated when the pump is in operation.

Early in the life of the unit this bearing gave trouble. The guide rings became broken consequently chewing up the collar and surfaces of the bearing plates. The guide rings were of very flimsy design. They had seventy-two 1/4" oil holes on a

11 9/16" diameter which brings them about 1/2" apart on centers. The rings were 1/2" thick and of about 1 3/8" section. Also there was such uneven wear on the plates that it was evident that either they were not serving their purpose properly or they were getting very unbalanced loading. Design shows that the former was not the case. The latter was probably caused partly by the lack of guide rings after same had become broken as mentioned above, and partly by uneven and unbalanced action of the revolving parts.

FUTILE ATTEMPTS AT CORRECTION OF FAULTS

At various times when the unit had undergone repairs, attempts were made to correct the faults. Evidently these were in the nature of making easy necessarily frequent remedial replacements of the same old design, rather than an effort at new design which would eliminate all trouble. With the guide rings in one piece the shaft would have to be lifted entirely clear of the bearing for replacement. The clever idea of making them in two parts was probably suggested by the dove-tail construction of the plates (see drawing). This of course, would permit the replacement of guide rings without taking out the shaft; but would not allow a press or tight fit on the collar without impractical complications.

Information is lacking as to the chronology of these various minor changes, but when the writer took hold of the work the collar, which originally was integral with the inside casting of the stationary half of the bearing, had been machined off

straight with the floor of the oil chamber and a bronze collar bolted in its place. There fortunately was enough room to permit this bronze collar to have a flange on it, so that it could be bolted down. This new collar was also made in two pieces. The job of making all the joints oil-tight at a time of replacement, must have been as hard as removing the split nut releasing other connections and letting the crane lift the shaft above the bearing.

If there were ten correct ways and one wrong method of putting a mechanism together, the so-called mechanics, found about a place of this sort, would be sure to choose the wrong one. The guide rings were not fitted tight to the collar; with the result that they revolved about it. Bronze against bronze is bad; the guide rings, having been made in halves were more flimsy than before; and when the bearing was opened, it was found to be a wreck.

INVESTIGATION OF BETTER DESIGN

While No. 1 unit was being overhauled an investigation was made with a view of trying to find a new design of bearing for the other three pumps. No. 1 had to be repaired and the bearing plates trued up, and reassembled as it had been, but there was time to consider other ideas for the remaining pumps. At first it was thought that the solution would be found in ball or roller bearings. Consultations were held with other engineers and designers who stood high in those fields but their rabble carried no convictions.

Ball and roller bearings of various types could be and were designed, but the inability to show them in practice on similar installations for a reasonable length of time and the lack of guarantees made them out of the question. An improvement was required, not an experiment, because, as has been said, continuous operation of a sewage pumping plant is absolutely essential.

Three qualifications of the new design presented

themselves as being necessary. They are, simplicity, elimination or correction of former troubles and durability. One is no more important than another. The first is required so that unskilled labor could not assemble the bearing wrong. The second explains itself. The third qualifies the materials and design entering into the work, and, although the second is accomplished, must preclude the possibility of adding new weaknesses. Ball or roller bearings are not simple in themselves and might entail complications in lubricating. To satisfy the second and third qualifications they would have to be a success in a similar installation. Success means uninterrupted service for a long time. Even then great care would have to be exercised in selection of bearings because where balls or rollers are a success often only one of many different and similar designs has proven applicable.

A plan of keeping the old design with minor changes was also considered. It was thought that

the guide rings could be made thicker radially necessitating, of course, a reduction of area of the bearing plates. But the bearing plates themselves were faulty. Even while this paper was being written one of the bronze plates on No. 3 unit became broken during a heavy rain. With No. 2 unit down it was necessary to develop makeshift repairs to get the unit back into service until a new plate could be procured from the factory. (From now on extra plates will be carried in the store room of the station).

Other schemes were suggested and considered but the best of all of them was the one made by the manufacturer of the unit, and the one which, after considerable commercial negotiation, was selected. Its simplicity at once evinces its correction of former troubles. That it is durable, the analysis of the design will show.

CHANGES DETERMINED UPON

The new thrust bearing consists of cast iron elements with a single pair of contact surfaces. (See plate 14). The collar wreckage has been replaced - as shown on drawing - by a one piece sleeve, pressed into the inside casting of the stationary half of the bearing.

The bearing is guided radially by the steady bearing, added to the lower face of the outside casting of the stationary half of the thrust bearing. The oiling system remains practically unchanged. Oil flows toward the shaft through radial grooves under the lower cast iron plate, rises in the space left for it at the center and is distributed over the bearing surfaces by the centrifugal action of tangential grooves. The perforated metal strip prevents the oil from being thrown by centrifugal force over the sides of the oil and plate chamber.

The steady bearing was a later development than the thrust bearing proper. Careful study of

the failure of the old mechanism showed that there must have been a whipping action of the shaft as it revolved. This action could have been independent of any wear in the Lignum-Vitae bearing. It was guided only by the main crank bearing at the top and by the Lignum Vitae bearing at the bottom. The distance between the two is more than 24 ft. and the shaft has two couplings within that space. Consequently it was decided that the guide bearing would be necessary to assure proper alignment centrally. With or without the steady bearing, it is manifest that no mechanical pressure can be brought to bear on the collar which forms the inside of the oil and bearing chamber.

The new parts were manufactured by the Allis-Chalmers Manufacturing Company, of Milwaukee - the makers of the pump. The split nut and inside casting of the stationary half of the bearing were shipped to the factory for the necessary machine work and fitting. The bearing was assembled by

employees of the Sanitary District. The reconstructed bearing is guaranteed by the Allis-Chalmers Manufacturing Company, to operate satisfactorily and that if it does not give service the new parts can be returned within one year and the money paid for them will be refunded. This guarantee, however, is contingent on the agreement of the Sanitary District to permit their expert to be at the plant when the unit was started for the first time after it was assembled, and to make any minor adjustments which he may see fit.

DESIGN OF NEW BEARING

It has long been known, and it is no secret that cast iron forms an excellent bearing for a cast iron journal. That the revolving half of this bearing is fundamentally a journal, needs no argument. In general, like metals make the poorest combination for bearings, but cast iron is one of the few exceptions. The coefficient of friction of cast iron against cast iron compares favorably with those of other bearing combinations and the surfaces of cast iron elements tend to wear together with increasing smoothness. In fact the faces of such bearings, after continued service resemble the surface of a piece of highly polished plate glass.

Other questions about the superiority of bearings of this structure and design were put to rest by the citation of tests on bearings of this type on similar units at a large hydraulic station. At this plant large units side by side were equipped with ball bearings, roller bearings the famous

Kingsbury bearing the old type thrust bearings and bearings of the style now under consideration. The units were all run in actual service under the same conditions, as nearly as possible. Among other tests, the rise of temperature of the lubricating oil was measured over various periods of time. The cast iron bearing and the Kingsbury showed about the same rise in temperature. This was less than for the other types and the net result of the tests, in general, was most favorable for the cast iron bearing. Needless to say that it was kept in service, and enough time has now elapsed since its original installation to prove its durability.

The Kingsbury bearing is a comparatively recent invention. It is much like the bearing here discussed but it differs in that the stationary disc is split up into segments. Each segment or shoe rests on a spherical seat; that is to say, the bottom of the segment is convex and the seat into which it fits is concave. This permits automatic com-

pensation for different rates of wear. Its design is based on the law that the wear is proportional to the speed, and the speed is greater near the outside of the disc than at the inside. It also satisfies the assumption that the film of lubricant is thickest at the point where it enters between the contact surfaces. It is more complicated than the bearing adopted for this unit; it costs more to make; and being a patented design it probably would involve royalty obligations.

The idea of using multiple C.I.or steel plates never was suggested, and if any of the numerous engineers, who were consulted on this work, ever thought of it before a decision was finally reached, he had conscience enough not to mention it. At any rate, inasmuch as the contact surfaces had to be designed with great enough area to permit the entire bearing action between a single pair of plates, the advantage gained by two heavier C.I.plates overwhelmingly outweighed

any arguments in favor of anything that would simulate the old flimsy construction.

For commercial purposes the design of the plates may be viewed from the following rough calculations, based on the assumption that the load is equally distributed over the bearing surfaces:

Weight of shaft, impeller, etc. 26000 lbs. = 13 tons

Dia. inlet at pump 42 inches

Dia. center impeller about shaft 10 inches

Effective area at inlet $\frac{\pi}{4}\left(\overline{42}^2 - \overline{10}^2\right) =$ 1307 sq. in.

Total head against which unit is designed to pump 24 ft.

Hydraulic Thrust

$$\frac{24 \times 62.5 \times 1307}{144} = 13614.6 \text{ lbs.} = 6.8 \text{ tons}$$

Total load carried by thrust bearing = wt. shaft, impeller, etc. + Hydraulic Thrust = 13 + 7 = 20 tons.

Outside dia. bearing surface = 25"

Inside dia. bearing surface = $12\frac{1}{4}$"

Area bearing surface $\frac{\pi}{4}\left(\overline{25}^2 - \overline{12\frac{1}{4}}^2\right)$ = 373 sq.in.

Average pressure on bearing surface:

40,000# ÷ 373 = 107 lbs. per sq. in.

An objection was raised by one of the engineers, to using a pressure as high as this. He was a man of considerable "marine" experience and made the point that on thrust bearings for ship propellers an average pressure of 80 lbs. per sq. in. was high and that 40 and even 30 were common.

Investigation revealed, however, that "marine" practice involves conditions unheard of in "stationary" design and although it was an hydraulic machine, nothing could be shown in the operation of this unit which should take it out of the "stationary" class. Furthermore, similar installations were operating successfully with Kingsbury bearings at pressures as high as 250# per sq. in. The inventor of this bearing claims that his bearing will

stand 450 lbs. per sq. in. without injury to the metal, and with high speeds, pressures as high as 5000 lbs. per sq. in. have been obtained experimentally with forced oiling.

A commercial method of determining a heating factor follows:

$$K = \frac{P \times n}{W}$$

where

K = heating factor
P = total load pounds
n = r.p.m.
W = width (radially) of bearing surface

Substituting in above, values for the bearing

$$K = \frac{40{,}000 \times 125}{6.125} = 816{,}327.$$

This is an arbitrary formula used by the Allis-Chalmers Company. Apparently it is based on the law that the wear at any point is proportional to the product of the pressure on, and velocity of the rubbing surfaces, at that point. It does not mean much in this form, except comparatively. Many bearings on their installations have a heating factor, according to this formula, of more than 1,000,000.

A more scientific check on this design follows:

Let R = radius of outside of disc or plate-inches

r = " of inside of disc or plate-inches

ϕ = normal pressure on bearing - lbs. per sq. in.

W = total pressure on bearing - lbs.

N = R. P. M.

μ = coefficient of friction

Mf = moment of friction - lb. inches

A = work of friction, per min. - ft. lbs. per min.

a = work of friction ft. lbs. per min. per sq. in. of projected area = $\frac{A}{\pi(R^2 - r^2)}$

q = heat generated B.T.U. per min. per sq. in. of projected area = $\frac{a}{778}$

Divide the face of the bearing into any number of concentric rings of equal width = dr.; that is, if any ring has an inside diameter of $2r_o$ its outside diameter is $2\left(r_o + dr\right)$. Then the area of any ring is:

$$2\pi r_1 \, dr$$

where r_1 = mean radius of the ring.

Then the total pressure on the ring is:

$$2\ p\ \pi\ r_1\ dr.$$

and the friction of the ring is:

$$2\ \mu\ p\ \pi\ r_1\ dr.$$

The moment of friction of the ring is:

$$d\ M_f = 2\ \mu\ p\ \pi\ r_1^2\ dr.$$

Then the moment of friction of the whole plate is:

$$\int d\ M_f = \int_r^R 2\ \mu\ p\ \pi\ r_1^2\ dr.$$

$$= 2\ \mu\ p\ \pi \int_r^R r_1^2\ dr.$$

$$M_f = 2\ \mu\ p\ \pi \left[\frac{r^3}{3}\right]_r^R$$

$$= \frac{2}{3}\ \mu\ p\ \pi \left(R^3 - r^3\right)$$

Total pressure is:

$$W = p\ \pi \left(R^2 - r^2\right)$$

Therefore(substituting):

$$M_f = \frac{2}{3}\ \mu\ W\ \frac{R^3 - r^3}{R^2 - r^2}$$

and:

$$A = \frac{2\pi N M_f}{12} = \frac{2 \times 2 \times \pi}{12 \times 3} N \mu W \frac{R^3 - r^3}{R^2 - r^2}$$

$$= .35 N \mu W \frac{R^3 - r^3}{R^2 - r^2}$$

Substituting:

$N = 125$ r.p.m.

$\mu = .02$ assumed

$W =$ 20 tons or 40,000 lbs.

$R = 12.5$

$r = 6.125$

$$A = .35 \times 125 \times .02 \times 40{,}000 \times \frac{\overline{12.5}^3 - \overline{6.125}^3}{\overline{12.5}^2 - \overline{6.125}^2}$$

$= 507{,}990$ ft. lbs. per min.

$$a = \frac{507{,}990}{\pi(R^2 - r^2)} = \frac{507{,}990}{373{,}016} = 1361.85 \text{ ft. lbs. per min. per sq. in.}$$

$$q = \frac{a}{778} = \frac{1361.85}{778} = 1.75 \text{ B.T.U. per min. per sq. in.}$$

This value would be high for an ordinary bearing, but inasmuch as it runs in a bath of oil and also is water cooled, it can be regarded as safe

design.

A similar check is applicable to the steady bearing. Instead, however, of "W" being an axial thrust as above it must here be considered as a radial load. Otherwise, except for the fact that the projected area, in this case, is at right angles (as shown below) to that of the thrust bearing, the principle is the same:

Let l = length of bearing - inches

d = diameter of journal - inches

other symbols as above

Then the projected area is ld

and

$$A = \frac{\pi d N W \mu}{12} \quad \text{ft. lbs. per min.}$$

$$a = \frac{A}{ld} = \frac{\pi N W \mu}{12 l}$$

$$q = \frac{a}{778} = \frac{\pi N W \mu}{778 \times 12 l}$$

$$= .0000067 \frac{N W}{l}$$

$$W = \frac{q l}{.0000067 N}$$

Substituting: $l = 12"$

$$W = \frac{12\ q}{.0000067 \times 125} = 14328.36\ q.$$

Assuming an average safe value for q = .6

$$W = 8597.02 \text{ lbs.}$$

$$p = \frac{W}{ld} = \frac{8597}{12x9} = 79.6 \text{ lbs. per sq. in. of projected area.}$$

To satisfy good design, therefore, the total radial pressure on the steady bearing should not greatly exceed ten thousand pounds. The purpose of this bearing, however, is not to carry a load but to steady the shaft. As long as the shaft never gets a chance to gallop or whip around, it is certain that the bearing will receive no such pressures. Should it be allowed to get badly worn its continued wear would become more rapid as time went on. The pressures involved under such conditions undoubtedly would be enormous, but theresulting damage is, in a measure, limited by the guide bearings near the top and bottom of the unit. The solution is to keep the steady bearing in good order, by proper lubrication and adjustment.

The greatest surface speed (at the outside of disc) of the thrust bearing proper is:

$$\frac{\pi N D}{12} = \frac{\pi \times 125 \times 25}{12} = 818.125 \text{ ft. per min.}$$

Similarly on the steady bearing the velocity is:

$$\frac{\pi \times 125 \times 9}{12} = 294.525 \text{ ft. per min.}$$

For the former, in accordance with pressures, oiling etc., 1000 ft. per min. would not be excessive. For the latter 500 ft. per minute is easily allowable.

CONCLUSION

In adopting a new bearing, it was said that there were three determining factors. These were simplicity, elimination or correction of former troubles, and durability. That the first has been attained needs only a glance at the blueprints for proof. That the second and third have been fulfilled do not necessarily remain to be seen. The mathematical delineation is the best prediction of what the bearing will do. All dimensions appear to be the result of careful design, comments upon which have already been made. There are no surface speeds which are out of proportion to conservative practice. In addition to keeping within reasonable limits in pressures, heating factors etc., the face width of the bearing surface has been kept narrow - only 6 3/8 inches. The advantage of this is that it eliminates too great a difference of surface speeds over the same area. The governing law states that the wear is faster at the outside of the plate than at the inside, in proportion to the product of the pressure on,

and the velocity of, the rubbing surfaces. It is probably this one attribute which caused this bearing to become such a keen competitor of the Kingsbury in the tests mentioned heretofore.

Many forms of bearing were considered and the only one which held any place near the chosen one, was the Kingsbury. This is more complicated, however, and its consequent higher cost excluded it. The others were rejected either because they were not proven in service or because their very form would not permit their removal and return to the old type, if they were not successful. An example of the latter case - and a very good bearing too - is the Schiele Curve type. Inasmuch as it pleased those in high authority to make this reconstruction under contract and guarantee, very little change was permissable in the surrounding mechanism, because in the event of failure, the new work has to come out and the old put back. The "Schiele" would have entailed a great deal of such change.

Structurally, there is nothing better for a

bearing surface than cast iron. Authorities agree that the actual contact of metal, with proper lubrication exists, for only about one quarter turn of the shaft at starting and stopping; the remaining time, it is believed that the load is carried on a film of oil.

Whatever contact there is, seems to improve the faces with use.

The cast iron used in these bearings is a special grade of very fine grain gray metal. The compresive strength of cast iron, being very large as compared to other materials used for bearings, is ample to withstand anything that is expected of it in this unit. The plates are of substantial thickness - one and seven eights (1 7/8) inches; in fact the top of the split nut had to be recessed to take care of the increased dimensions.

The use of cast iron here has a further meaning. Of late years many articles formerly made of cast iron, especially those involving com-

plicated stresses, have been made of steel. The engineering profession has been inclined to consider cast iron of value for only base plates, foundations, engine beds, and the like. It is human to jump from one thing to another without exhausting ones resources. A few engineers, however, have clung to a technical evolution of cast iron and are now showing that it holds an important position with scientifically employed metals. To verify this statement the reader has only to investigate the chilled iron R. R. car wheel. Car wheels have been tried in steels, but as in many other attempts at betterment, the manufacturers have been forced to admit the superiority of cast iron for the particular purpose, and have turned back to its improvement. Cast iron some day will probably take its place among the highly developed metals.

The analysis of the new design indicates in every respect an accomplishment of purpose. The bearing as reconstructed is so simple that it is almost impossible to assemble it incorrectly. The

lubrication is simple and positive. Former troubles are overcome. No pressure reaches the collar. There are no guide rings or other collection of small parts of flimsy construction. The design seems inherently to be free from any similar new troubles. The sizes of all parts are commensurate, with the service that is expected of them, and are ample to insure durability.

REFERENCES

H. L. Nachman:

"Elements of Machine Design"

F. A. Halsey

"Handbook for Machinists,Shop Men and Draftsmen"

Henry J. Spooner

"Machine Design, Construction and Drawing"

Lionel S. Marks

"Mechanical Engineers Hand-Book"

L. P. Alford

"Bearings and Their Lubrication"

PLATE I2.

Assembly Drawing of 75 C. F. S. Sewage Pumping Unit.

THESIS.

Reconstruction of a Thrust Bearing for a 75 C. F. S.
Centrifugal Sewage Pumping Unit.

By

L. W. A. BUNGE.

CLEARANCE SHEET - CLEARANCE

75 CUFT CENTRIFUGAL PUMPING [illegible]

12-22" 32 x 30

39TH ST STATION

CHICAGO ILL

Thrust Bearing

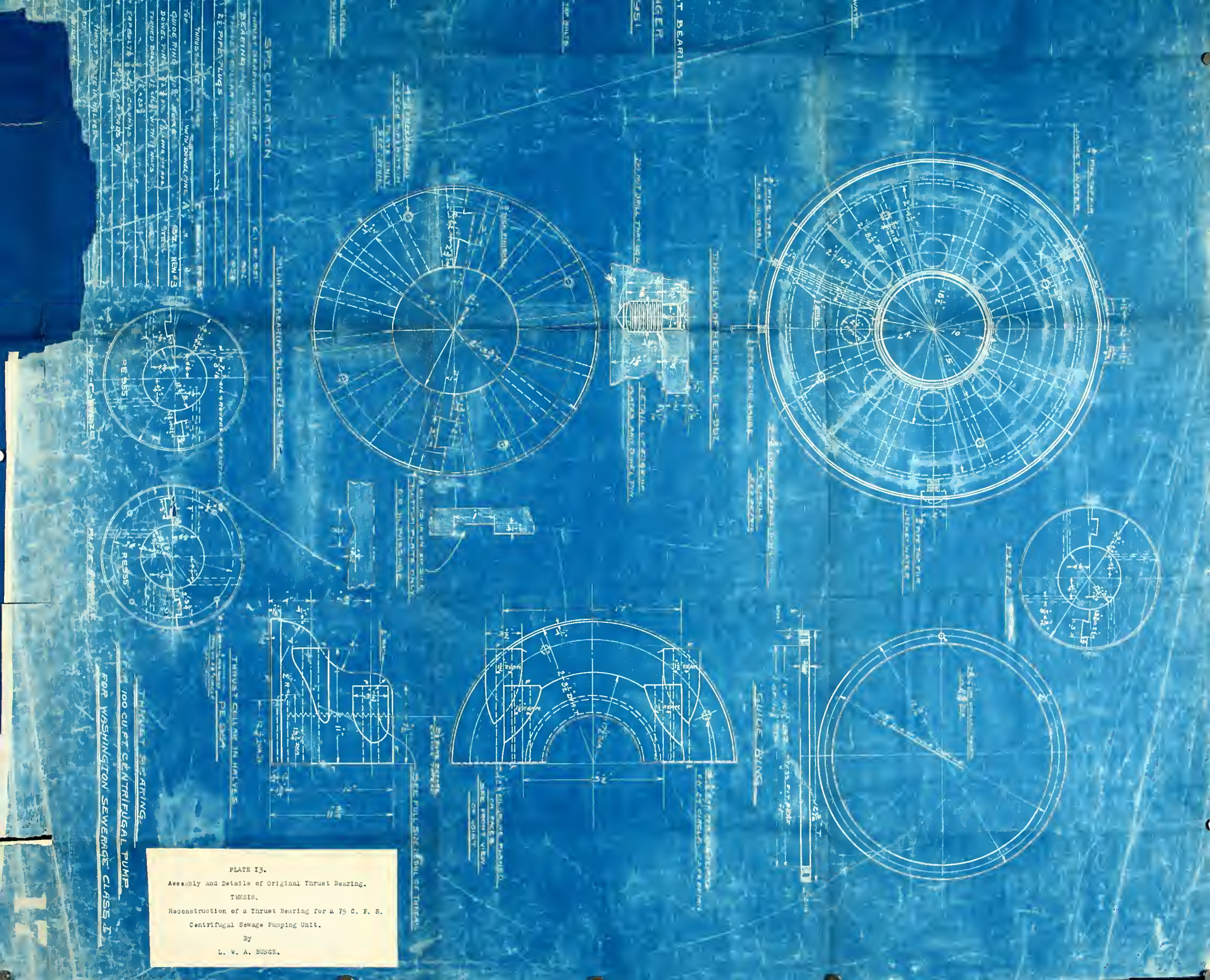


PLATE I3.

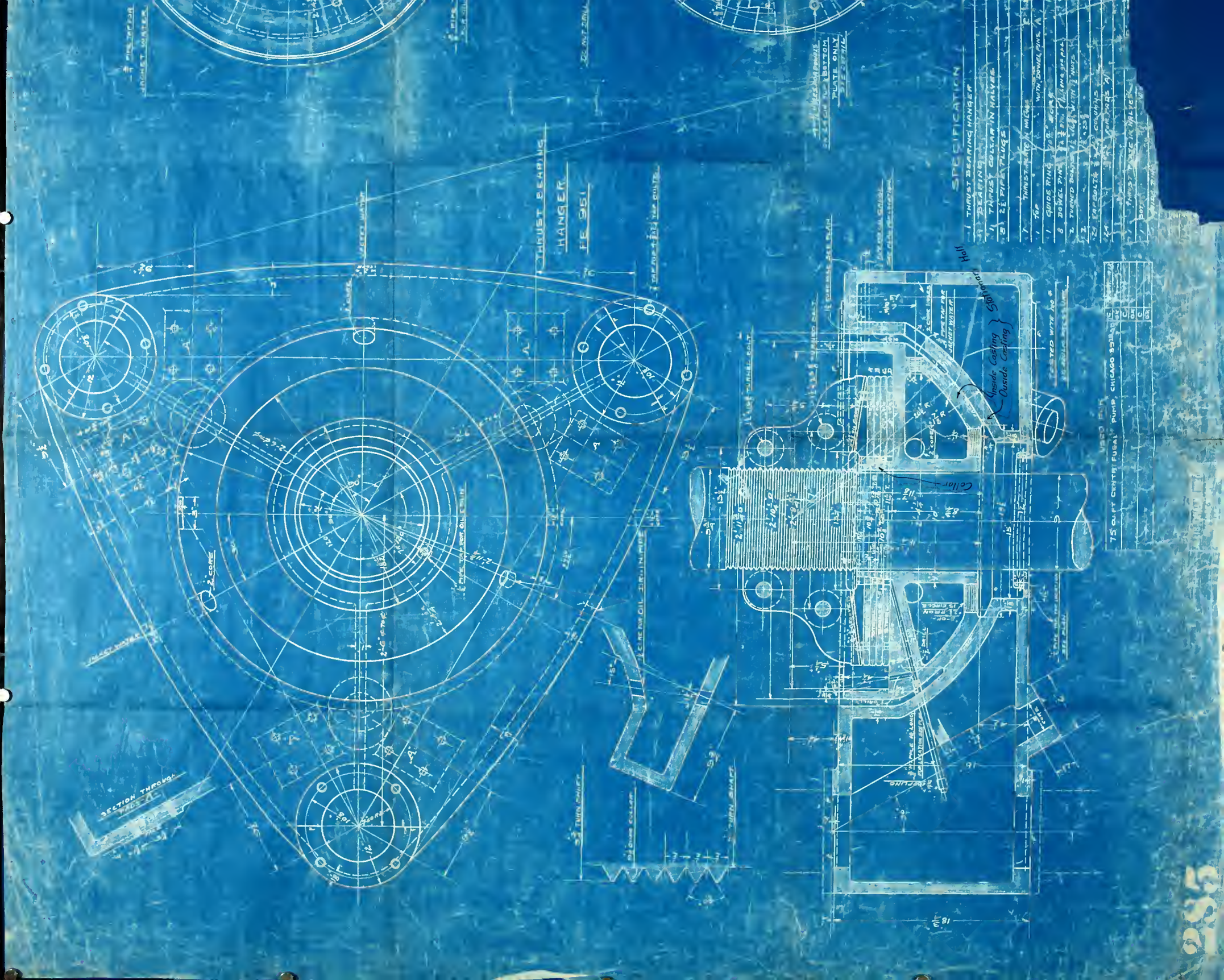
Assembly and Details of Original Thrust Bearing.

THESIS.

Reconstruction of a Thrust Bearing for a 75 C. F. S.
Centrifugal Sewage Pumping Unit.

By

L. W. A. BUNGE.


THRUST BEARING HANGER
FE 951
SPECIFICATION
1 THRUST BEARING HANGER
1 BEARING
1 THRUST COLUMN IN HALVES
8 2½ PIPE PLUGS
1 TOP THRUST HOUSING
1 GUIDE RING
8 DOWEL PINS
2 TURNED BOLTS
24 FAROLTS
SECTION THROUGH
Inside Casting
Outside Casting
Collar
75 GPM CENTRIFUGAL PUMP CHICAGO

Used For		№	Ord.№
39th St. Sta. Sanitary Dist. City of Chicago	Unit 1 of 15		I-H-1177

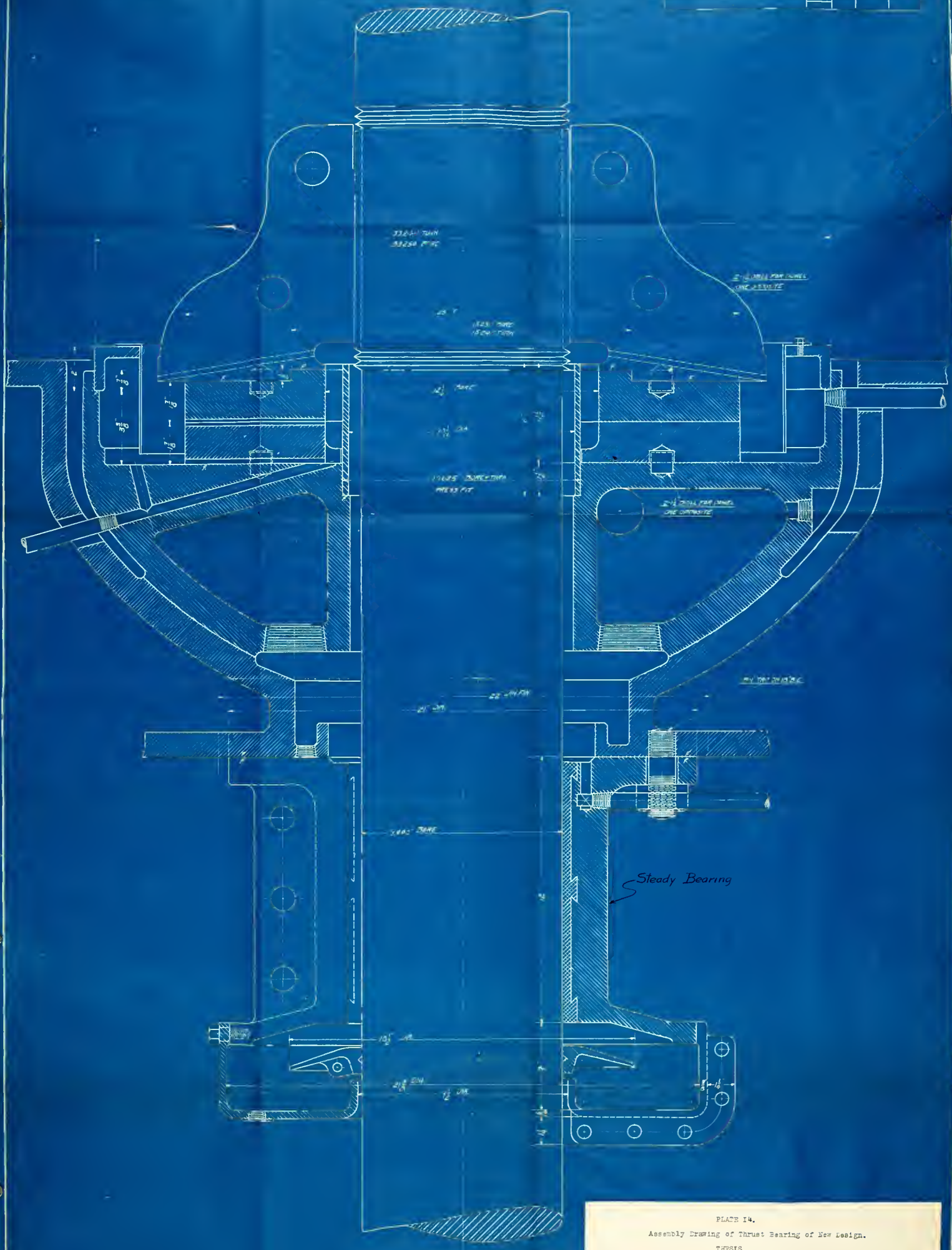


PLATE I4.

Assembly Drawing of Thrust Bearing of New Design.

THESIS.

Reconstruction of a Thrust Bearing for a 75 C. F. S.

Centrifugal Sewage Pumping Unit.

By

L.W.A. BUNGE.

1	4-17-12	ALLIS-CHALMERS MFG. CO.	Thrust-Steady Bearing	929
			for	—
			15 C.F. Vertical Cent. Pump	15